NISTIR 6226

Performance of a Commercial Hot Water Boiler

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Abstract

In responding to ASHRAE SPC 155P's interest and to provide additional evidence on a possible linear relationship between output and input of a boiler under part load conditions, the National Institute of Standards and Technology undertook to conduct a series of computer simulations utilizing a revised version of a computer simulation program, DEPAB2. This program was development previously by NIST for commercial fire-tube steam boilers, and was modified so that it could be used to simulate the operations of a commercial fire tube hot water boiler. Actual boiler specification data from a leading boiler manufacturer was used as input data to the program for the simulation study.

In this paper, the computer model of a commercial hot water boiler and the actual boiler specification are described. Full and part-load performance of the simulated, commercial hot water boiler is presented. A good linear correlation between output energy and input energy is found. Based upon the correlation, a suggestion is made to reduce experimental burden in testing the performance of commercial boilers.

Keywords

boiler, boiler computer model, boiler part-load performance, commercial boiler, fire-tube boiler, hot water boiler

ACKNOWLEDGMENT

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1. Introduction

In an effort to conserve energy, a test procedure for residential space heating boilers and furnaces [1, 2] was developed and adopted by the U.S. Department of Energy (DOE) to determine the heating seasonal efficiency of these products. At the present time, however, there is no suitable test procedure for commercial boilers which takes into account part-load operations. The American Society of Heating, Refrigerating, and Air-conditioning Engineers, Inc. (ASHRAE) Standard Project Committee (SPC) 155P has been working on development of such a test procedure. Since the conducting of the tests under part loads operating conditions is usually a costly and time-consuming task, the development of a simpler, yet accurate test method for determining the part load efficiencies of commercial boilers is one of the major concerns of the ASHRAE SPC 155P and the Department of Energy.

During the course of deliberations by the ASHRAE SPC 155P on the part load performance of commercial boilers, field test data collected from previous studies by the Center for Energy and Environment, Minneapolis, MN [3] were examined which indicated that the output and input of boilers could be correlated by a straight line approximation. That is, the output of a boiler under part load operation was found to be approximately a linear function of its input. The SPC 155P felt that if this linear correlation hold true for different types of boilers, such as cast iron boilers, steel boilers, etc., the number of tests required to arrive at a part load efficiency curve could be greatly reduced.

In order to assist the ASHRAE SPC 155P Committee and to provide additional information on the possible linear relationship between output and input of a boiler under part load conditions, the National Institute of Standards and Technology undertook to conduct a series of computer simulations utilizing a revised version of the boiler simulation program, DEPAB2 [4]. This computer program which was developed previously by NIST for commercial fire-tube steam boilers was modified so that it could be used to simulate the operations of a commercial fire tube hot water boiler. Actual boiler specification data from a leading boiler manufacturer was obtained and used as input data to the program for the simulation study.

In this paper, the computer model of a commercial hot water boiler and the actual boiler specifications are described. Full and part-load performance of the simulated commercial hot water boiler is presented. A good correlation between output energy and input energy is found. Based upon the correlation, a suggestion is made to reduce experimental burden in testing the part load performance of commercial boilers.

2. Computer Model of a Commercial Water Boiler

The NIST DEPAB2 program is a computer simulation program for the DEsign and Performance Analysis of commercial Boilers [4]. The program, which is capable of

simulate fire-tube boilers, contains the routines for simulating boiler controls, radiative/convective heat transfer in the boiler combustion chamber and boiler tubes, waterside heat transfer, and boiler shell losses.

Since the DEPAB2 simulated only commercial steam boilers, it was necessary to substantially modify the program in order to simulate water boilers. The code written in Fortran IV was converted into Fortran 77 and many lines of code for water boilers were added. In addition, a part-load performance evaluation capability was included and the input/output data formats were changed for clarity. The newly modified program is now called DEPAB2R. All the simulation runs presented here were made using this DEPAB2R.

Although DEPAB2R can be used for either steam boilers or water boilers, only the results from water boiler simulation will be described in this paper. Figure 1 is a schematic view of a typical hot water commercial boiler with three sequential gas paths.

2. 1 Boiler controllers

For boiler controls, on/off, high/low/off, and modulating controllers are considered. The boiler water temperature is sensed by a water temperature sensor that is modeled as a first order differential equation given by:

$$\tau_{sen} \frac{dT_{sen}}{dt} + T_{sen} = T_{bw}, \tag{1}$$

where T_{sen}, T_{bw} and τ_{sen} are the sensor temperature, the boiler water temperature, and the time constant of the sensor, respectively.

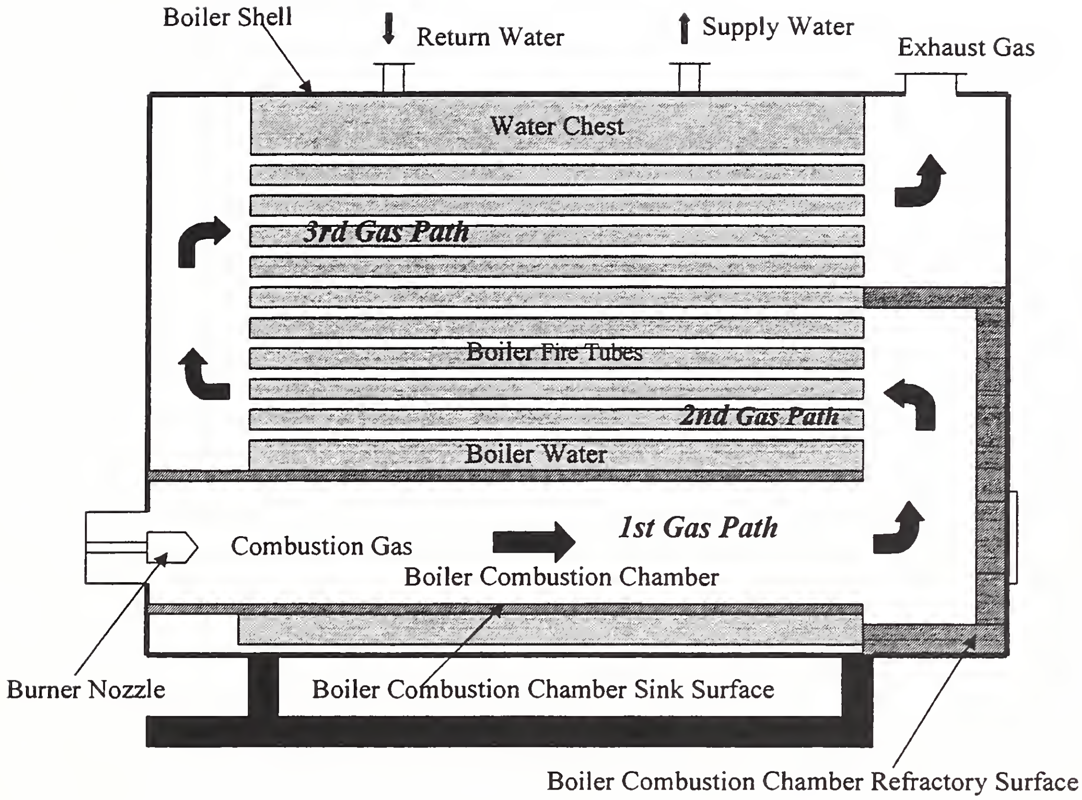

Figure 1. Schematic diagram of a typical hot water commercial boiler with three paths.

The on/off controller (single-stage controller) has the simplest control logic among the three controllers. If the sensor temperature is less than a low limit setpoint, the controller turns the boiler on. If the sensor temperature is greater than a high limit setpoint, the controller turns the boiler off. The control signal controls the fuel and airflow for the boiler system.

The fuel and airflow rates are at the maximum values, when the control signal is "on". But when the control signal is "off", the fuel flow stops and the air flow rate through the boiler is assumed to be governed by the equation:

$$\tau_{air} \frac{d\dot{m}_{air,off}}{dt} + \dot{m}_{air,off} = \dot{m}_{air,\min}, \tag{2}$$

where $\dot{m}_{air,off}$, $\dot{m}_{air,\min}$, and τ_{air} are the off-cycle air mass flow rate, the minimum airflow rate, and the airflow time constant, respectively.

When the high/low/off controller (two-stage controller) is used, three temperatures must be specified. These are the high limit temperature setpoint, the low limit setpoint, and the switchover temperature. If the sensor temperature is lower than the low limit setpoint, the

fuel flow is at the high firing rate and the airflow is also at its full rate value. If the sensor temperature is higher than the high limit value, the controller signal for the fuel calls for no fuel and the airflow is modeled by equation (1). The low firing rate value is used when the sensor temperature is greater than the switch-over setpoint and below that the high limit temperature setpoint, while the high firing rate value is used, if the sensor temperature is lower than the switch-over setpoint and above the low limit temperature setpoint. At the low firing rate, the airflow rate is also reduced to the lower value.

The air/fuel metering controller (modulating controller) operates in a similar way as the high/low/off controller (two-stage controller) except that the fuel flow as well as the air flow are modulated following a specific, tabulated pattern in the temperature range where the two-stage control calls for full flow rates.

2.2 *Boiler combustion chamber heat transfer*

During the burner on-period, the heat transfer between the combustion gas and the combustion chamber surface of the boiler due to convection and radiation is based upon the stirred reactor theory [5], in which the entering air and fuel are assumed to have sufficient momentum to assure a reasonably well-stirred combustion chamber. The radiative and convective heat flow from combustion gas to the boiler combustion chamber surface, $\dot{Q}_{f,on}$, is given by:

$$\dot{Q}_{f,on} = \left[A_{gs} \left(\frac{1-K^3}{1-K^4} \right) + A_s \frac{2h_c}{\sigma \left(T_{ag} + T_s \right)^3} \right] \sigma \left(T_{ag}{}^4 - T_s{}^4 \right), \tag{3}$$

where

$$A_{gs} = \frac{A_s + A_r}{\dfrac{1}{\varepsilon_g} + \dfrac{1}{C_s \varepsilon_s} - 1}, \quad C_s = \frac{A_s}{A_s + A_r}, \quad K = \frac{T_s}{T_{ag}}, \text{ and } T_{ag} = \frac{1}{T_{af} + T_{eg}}. \tag{4}$$

A_r : combustion chamber refractory surface area
A_s : combustion chamber sink surface area
h_c : convective heat transfer coefficient
T_{af} : absolute temperature of adiabatic flame
T_{eg} : absolute temperature of combustion gas exiting from combustion chamber
T_s : absolute temperature of combustion chamber sink surface
ε_g : combustion gas emissivity
ε_s : combustion chamber sink surface emissivity
σ : Stefan-Boltzmann constant (5.67 x 10^{-11})

The convective heat transfer coefficient, h_c, is obtained from McAdam's correlation equation [6]:

$$h_c = 0.023\frac{k}{D_h}\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4},\quad \mathrm{Re} > 2000, \tag{5}$$

where k, D_h, Re, and Pr are the thermal conductivity, the hydraulic diameter of the boiler combustion chamber, the Reynolds number, and the Prantl number, respectively.

During the burner off-period, "draft air" flows through the boiler combustion chamber. Since the draft air can be considered radiatively transparent, the heat transfer is pure convective. Applying the effectiveness method for a compact heat exchanger [7], the off-cycle heat flow rate, $\dot{Q}_{f,off}$, is given by:

$$\dot{Q}_{f,off} = c_{pa}\dot{m}_{air,off}\left(T_a - T_{bw}\right)\left(1 - e^{-NTU}\right), \tag{6}$$

where

$$NTU = \frac{A_s h_a}{c_{pa}\dot{m}_{air,off}}. \tag{7}$$

NTU is the number of heat transfer units, c_{pa} is the specific heat of air, T_a is the boiler room ambient air temperature, and T_{bw} is the boiler water temperature, respectively. The quantity h_a is the convective heat transfer coefficient given by:

$$h_a = a\frac{k}{D_h}\mathrm{Re}^{b}\,\mathrm{Pr}^{c}, \tag{8}$$

where a, b and c are constants.

2.3 Boiler heat-exchanger tube heat transfer

The boiler heat-exchanger tubes in commercial boilers can be divided into elementary sections in the gas flow direction. For each elementary segment, the net radiative heat flux in the direction of gas flow can be ignored relative to the flux normal to the gas flow direction. The heat transfer between the gas and the tube wall can be evaluated using the long-tube analysis [5]. Thus the radiative heat transfer coefficient, h_r, is:

$$h_r = \frac{4\sigma \overline{T}_{gt}^{\,3}}{\frac{1}{\varepsilon_g} + \frac{1}{\varepsilon_t} - 1}, \tag{9}$$

where $\overline{T}_{gt}$, ε_g, and ε_t are the average of inlet and exit gas temperatures, the gas emissivity, and the tube wall emissivity, respectively.

The gas-to-tube convective heat transfer coefficient has the same form of equation as the combustion chamber heat transfer coefficient. The overall heat transfer coefficient, h_{all}, is defined as the sum of convective and radiative heat transfer coefficients.

The heat flow rate at the gas/tube interface of the i-th tube segment during the on-cycle, $\dot{Q}_{t,on}$, can be obtained from:

$$\dot{Q}_{t,on} = c_{pg}\dot{m}_{gas}\left(T_{g,in} - T_{ts}\right)\left(1 - e^{-NTU'}\right), \tag{10}$$

where

$$NTU' = \frac{A_t h_{all}}{c_{pg}\dot{m}_{gas}} \tag{11}$$

c_{pg}: gas specific heat
$\dot{m}_{gas}$: gas mass flow rate
$T_{g,in}$: gas temperature of the inlet of tube segment
T_{ts}: tube wall surface temperature
A_t: surface area of tube segment

The heat flow rate due to the draft airflow during the burner off-period for the i-th tube segment is given by:

$$\dot{Q}_{t,off} = c_{pa}\dot{m}_{air,off}\left(T_{a,in} - T_{ts}\right)\left(1 - e^{-NTU''}\right), \tag{12}$$

where

$$NTU'' = \frac{A_t h'_a}{c_{pa}\dot{m}_{air,off}}. \tag{13}$$

$T_{a,in}$ is the air temperature at the inlet of the tube segment and h'_a is the convective heat transfer coefficient that has the same form as for the combustion chamber except that its characteristic length is based on the tube diameter.

2.4 Heat transfer from heat-exchanger tubes to water

Heat exchangers of commercial boilers usually consist of banks of fire-tubes. The convective heat transfer from the walls of tubes to the water flow across tube-banks that are 10 or more rows deep is expressed by the correlation [7]:

$$Nu = C\,\mathrm{Re}^{0.6}\,\mathrm{Pr}^{0.333} \qquad \text{for Re} > 6000, \tag{14}$$

where Nu, Re, and Pr are Nusselt, Reynolds, and Prantl numbers based on the tube diameter and the maximum water flow speed through the gap between tubes. The constant, C, is 0.33 for staggered tubes and 0.26 for in-line tubes.

2.5 Shell heat transfer

Heat transfer through the boiler shell from the boiler water to ambient air is determined from the overall heat transfer coefficient, U_{shell}, the shell surface area, A_{shell}, and the temperature difference of boiler water and ambient air across the shell.

$$\dot{Q}_{shell} = A_{shell} U_{shell} \left(T_{bw} - T_a\right) \tag{15}$$

2.6 Heat balance of water boilers

Fuel with a higher heating value of HHV, specific heat $c_{p,fuel}$, fuel temperature T_{fuel} flows to the boiler burner with mass flow rate $\dot{m}_{fuel}$. The fuel burns to produce the input heat flow rate $\dot{Q}_{input}$:

$$\dot{Q}_{input} = \dot{m}_{fuel}\left[HHV + c_{p,fuel}\left(T_{fuel} - T_a\right)\right]. \tag{16}$$

During the on-period, the delivered heat from combustion gas to boiler water is equal to the heat flow rate from combustion chamber and all the segments of heat-exchanger tubes to water, $\dot{Q}_{total}$.

$$\dot{Q}_{total} = \dot{Q}_{f,on} + \sum_{i=1}^{n_t} \dot{Q}_{t,on,i}\,, \tag{17}$$

where $\dot{Q}_{t,on,i}$ and n_t are the heat transfer rate from the gas to the tube wall of the i-th tube segment and the number of tube segments of the tube-banks, respectively.

The combustion gas loses heat energy due to being exhausted to the outside and from the latent energy carried away by the exiting flue gases. Since latent heat loss, $\dot{Q}_{lat}$, can be

obtained from fuel properties, the exhaust heat loss, $\dot{Q}_{ext}$, is determined from heat balance equation:

$$\dot{Q}_{ext} = \dot{Q}_{input} - \dot{Q}_{total} - \dot{Q}_{lat} . \tag{18}$$

Knowing that the net heat gain in boiler water, $\dot{Q}_{net}$, is equal to $\dot{Q}_{total} - \dot{Q}_{shell}$, the water-side heat balance is given by:

$$\dot{Q}_{net} = \dot{Q}_{input} - \dot{Q}_{lat} - \dot{Q}_{ext} - \dot{Q}_{shell} . \tag{19}$$

The load, $\dot{Q}_{load}$, can be expressed by:

$$\dot{Q}_{load} = \dot{m}_w \left(c_{pw,s} T_{bw} - c_{pw,r} T_{rw} \right), \tag{20}$$

where $\dot{m}_w$, $c_{pw,s}$, and $c_{pw,r}$ are the circulating water mass flow rate, the specific heats of water at the supply port and at the return port with temperature T_{rw}. It is assumed that the boiler water is of a uniform temperature.

At a transient condition, the difference of net heat flow and load increases or decreases the boiler temperature as follows:

$$C_{tot} \frac{dT_{bw}}{dt} = \dot{Q}_{net} - \dot{Q}_{load} , \tag{21}$$

where C_{tot} is the sum of all capacitances of the boiler including boiler water, shell, heat-exchanger tubes, and combustion chamber.

2.7 Fuel utilization efficiency

At any instance of time, the ratio of load $\dot{Q}_{load}$ to fuel input rate $\dot{Q}_{input}$ is defined as instantaneous fuel utilization efficiency, η_t. When the boiler operation reaches steady state, the instantaneous efficiency becomes the steady-state fuel utilization efficiency, η_{ss}. At part-load conditions, the boiler controller operates in a cyclic mode. Depending upon load conditions, the controller turns on or off the burner and air pumps. During the off-cycle, $\dot{Q}_{input}$ is equal to zero. The cyclic fuel utilization efficiency can thus be expressed in terms of the boiler on-time t_{on} and the boiler off-time t_{off}:

$$\eta_{cyc} = \frac{\int_0^{t_{on}+t_{off}} \dot{Q}_{load} dt}{\int_0^{t_{on}} \dot{Q}_{input} dt} = \frac{E_{out,cyc}}{E_{inp,cyc}} , \tag{22}$$

where $E_{out,cyc}$ is output energy per cycle, and $E_{inp,cyc}$ is input energy per cycle.

3. Description of the Simulated Boiler

The input data to DEPAB2R program was generated, based upon an actual commercial steam boiler design data supplied by one of leading boiler manufacturers.

The boiler is a 4,905 kW (500 BHP), 3-pass, cylindrical type boiler. The inner diameter and length of the combustion chamber are 1.09 m (42.91 in) and 3.0 m (118.11 in), respectively. In the 2nd pass of combustion gas, there are 206 heat-exchanger fire-tubes of which fireside heating surface area is 110.79 m^2 (171,724.84 in^2). There are 156 fire-tubes with fireside surface area of 102.81 m^2 (159,355.82 in^2) in the 3rd pass.

It is assumed that the 2.31 m^3 (81.58 ft^3) chest of the boiler is filled with water. The total water volume contained in the simulated water boiler thus becomes 10.81 m^3 (381.75 ft^3). The 14.3 mm (0.56 in) thick steel shell with the outside diameter of 2.62 m (103.15 in) weighs 3,710.39 kg (8,180 lb).

It is assumed that the circulating water pump operates continuously with a relationship between the circulating water mass flow rates and temperature rises across the boiler system given in Figure 2 for full load conditions. The figure was plotted using tabulated values of a 500 BHP boiler in the boiler manufacturer's manual.

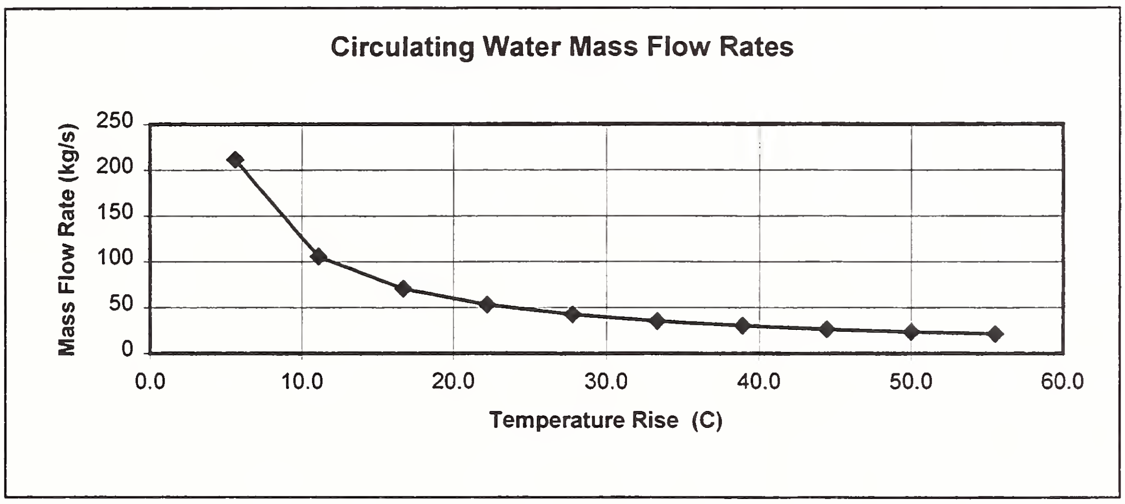

Figure 2. Circulating water mass flow rates with respect to system temperature rises of a typical real boiler.

4. Performance of the Simulated Commercial Water Boiler

Steady state and part-load performances of the water boiler were evaluated through a series of computer simulation runs. In these simulations, the on/off controller controlled the boiler operations. Both upper and lower limits for the controller were changed as needed.

4.1 Steady-state performance

Figure 3 shows the boiler water temperature with respect to time. When the return water temperature is $60.0\,°C$ ($140.0\,°F$), the circulating water mass flow rate is 53.0 kg/s (42064.2 lb/h). The boiler water temperature reaches at $82.6\,°C$ ($180.68\,°F$) at steady state. This gives a temperature rise of $22.6\,°C$ ($40.68\,°F$), which is very close to the relation of a typical real boiler shown in Figure 2. As shown in Figure 4, the instantaneous fuel utilization efficiency, η_t, approaches a constant value of 0.804 which is the steady state fuel utilization efficiency, η_{ss}, as time increases.

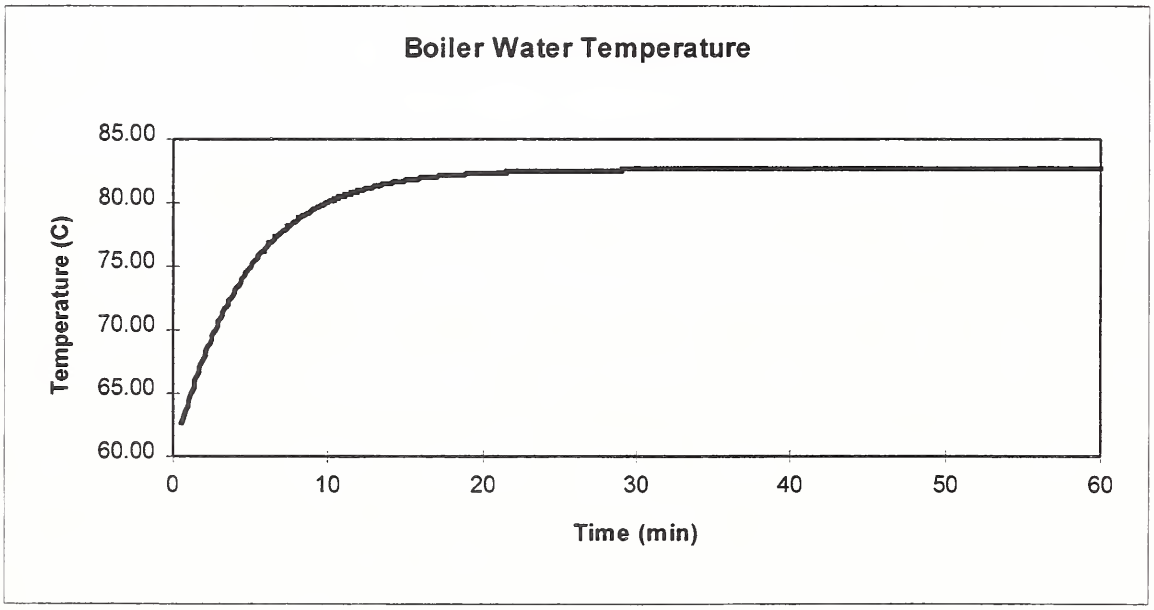

Figure 3. Boiler water temperature with respect to time.

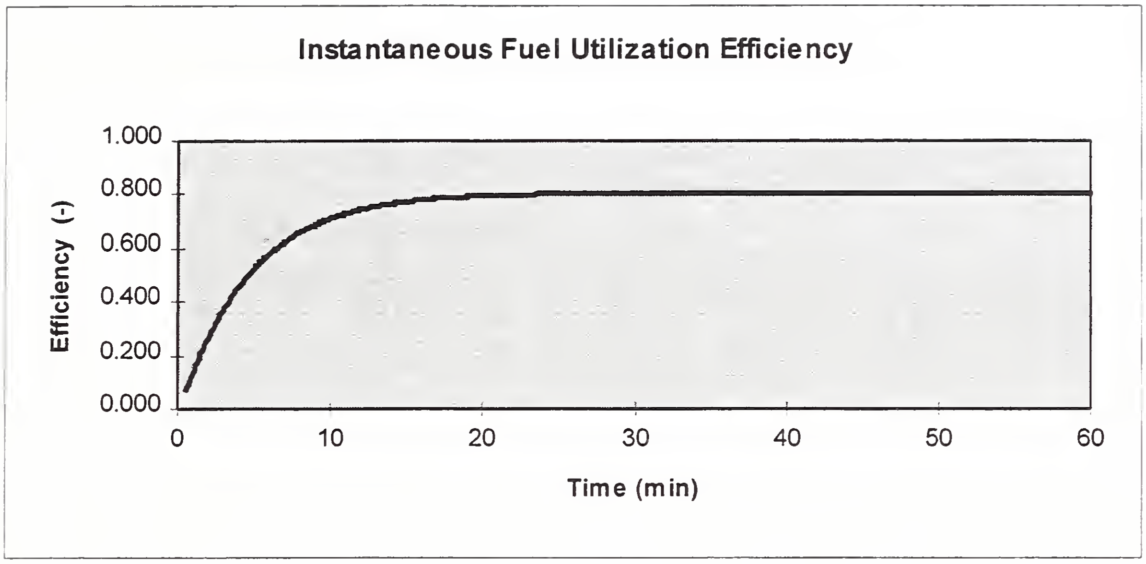

Figure 4. Instantaneous fuel utilization efficiency in terms of time.

Steady-state efficiency changes with respect to temperature rises, $T_{bw} - T_{rw}$, across the boiler system are shown in Figure 5. It can be seen that the higher the temperature rise is, the lower the steady-state efficiency is. In other words, a higher water flow rate results in higher steady-state fuel utilization efficiency.

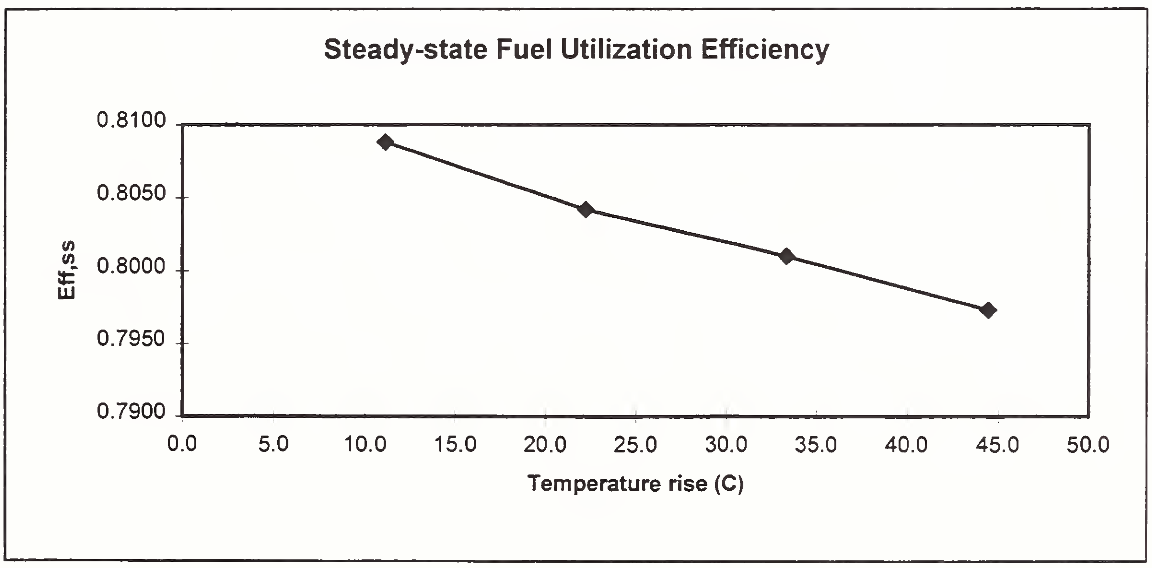

Figure 5. Steady-state fuel utilization efficiency, $E_{ff,ss}$, with respect to system temperature rise.

4.2 Part-load performance

The circulating water is assumed to circulate continuously through the boiler system regardless of load conditions. By using a three-way valve, a part of the circulating water is delivered to meet the load requirement under a part-load condition; the remainder of the water returns to the boiler. In the stand-by mode (no load condition), the circulating water returns back to the inlet of the boiler with almost the same temperature as the supply water temperature. Under the assumption that the boiler water is fully mixed, the supply water at the outlet of the boiler is the same as the boiler water temperature.

In this study, the setpoint of the upper limit of the boiler on/off controller is set to $82.22\,°C\,(180.0\,°F)$. When the boiler temperature reaches this upper limit, the boiler burner is turned off and the operation of the air pump for combustion air is set at the minimum level. To simulate part-load operation, the lower setpoint was varied. It should be noted that actual boiler controllers have a much narrower band of setpoint limits. The lower the low setpoint value is, the longer the boiler on-period is.

A part-load simulation was performed with the circulating water mass flow rate of 53.0 kg/s (42064 lb/h), corresponding to a temperature rise of $22.22\,°C\,(40.0\,°F)$, and the results are shown in Figures 6 through 9. The lower setpoint of the controller is $60\,°C$ $(140.0\,°F)$ above this temperature which the boiler burner is turned on and its air pump operates at the maximum level to provide the full amount of combustion air.

Figure 6 shows the normalized percentage input heat flow rates and the normalized percentage delivered heat flow rate to the load, Q^*_{load}. The normalized percentage exhaust heat flow rate, Q^*_{ext}, and the normalized percentage latent heat flow rate, Q^*_{lat}, are shown in Figure 7. All are normalized by the steady-state input heat flow rate, $\dot{Q}_{inp,ss}$.

$$Q^*_{inp} = \frac{100\dot{Q}_{inp}}{\dot{Q}_{inp,ss}}, \quad Q^*_{load} = \frac{100\dot{Q}_{load}}{\dot{Q}_{inp,ss}}, \quad Q^*_{ext} = \frac{100\dot{Q}_{ext}}{\dot{Q}_{inp,ss}}, \text{ and } Q^*_{lat} = \frac{100\dot{Q}_{lat}}{\dot{Q}_{inp,ss}} \tag{23}$$

A very small amount of heat is lost through the boiler shell as shown in Figure 8. Q^*_{shell} is the percentage heat loss rate through the boiler shell as defined by:

$$Q^*_{shell} = \frac{100\dot{Q}_{shell}}{\dot{Q}_{inp,ss}}. \tag{24}$$

Figure 9 gives the boiler water temperature variation with respect to time, resulting from the cyclic operation of the boiler. From the figure, it can be seen that the boiler water temperature varies in a wide range due to the lower setting of the controller. The controller low limit setting was purposely lowered to simulate part-load conditions.

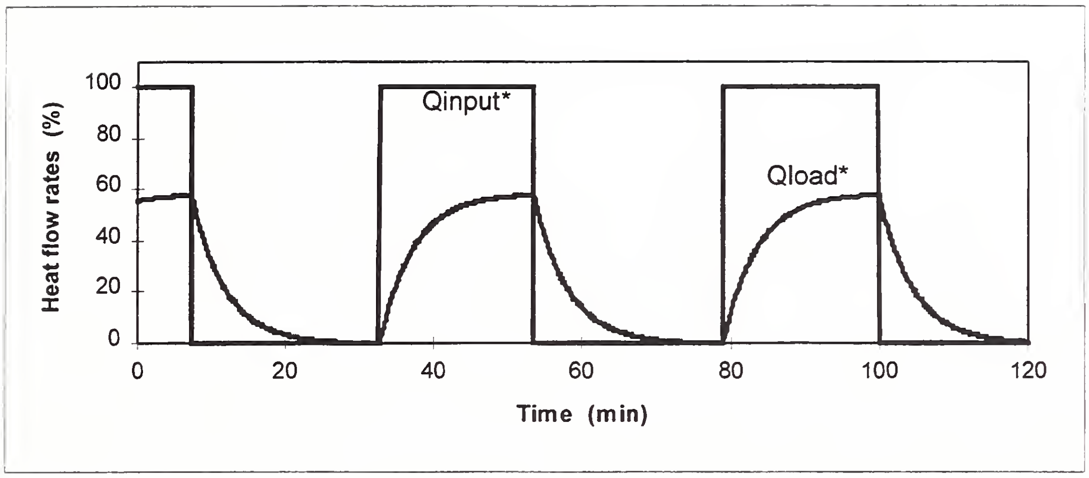

Figure 6. Normalized percentage input heat flow rate and delivered heat flow rate to the load.

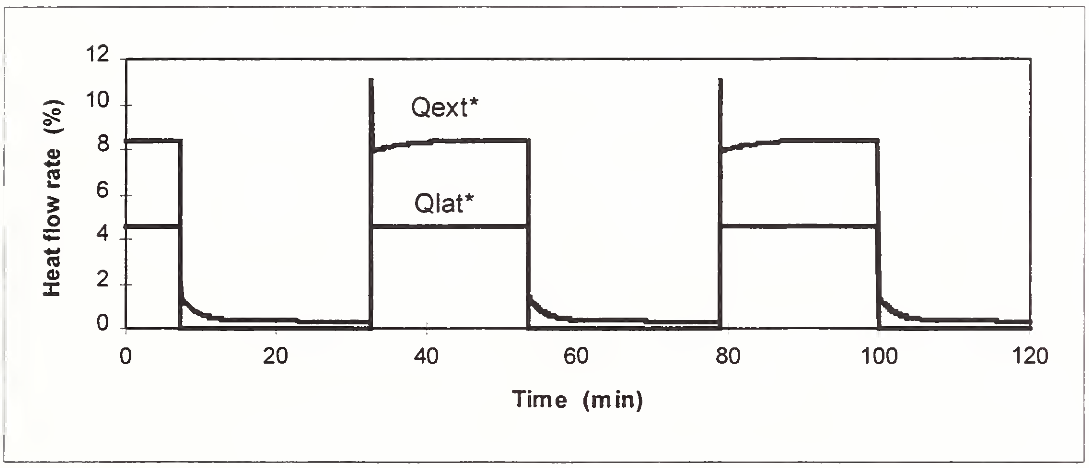

Figure 7. Normalized percentage exhaust heat flow rate and latent heat flow rate.

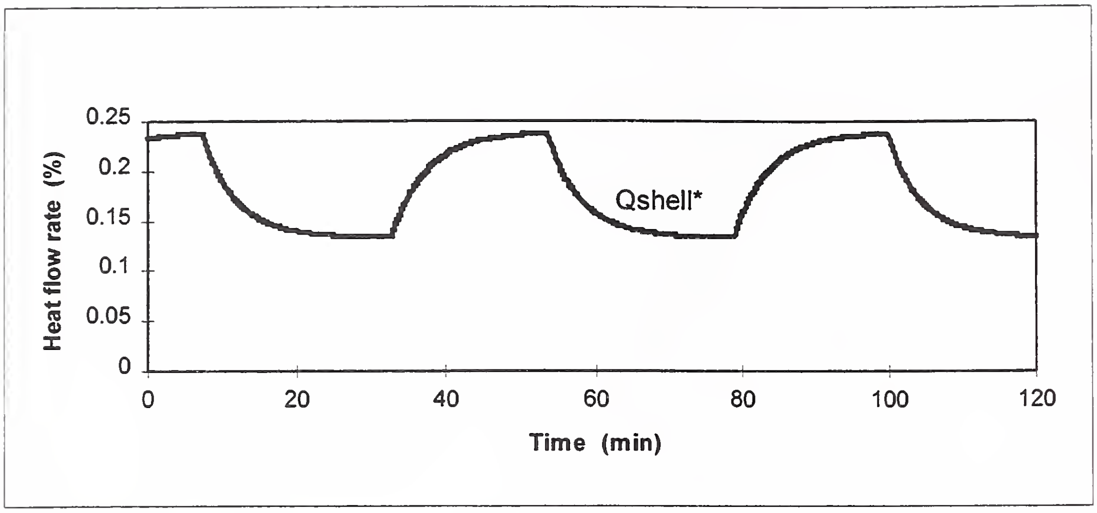

Figure 8. Normalized percentage heat loss rate through the boiler shell.

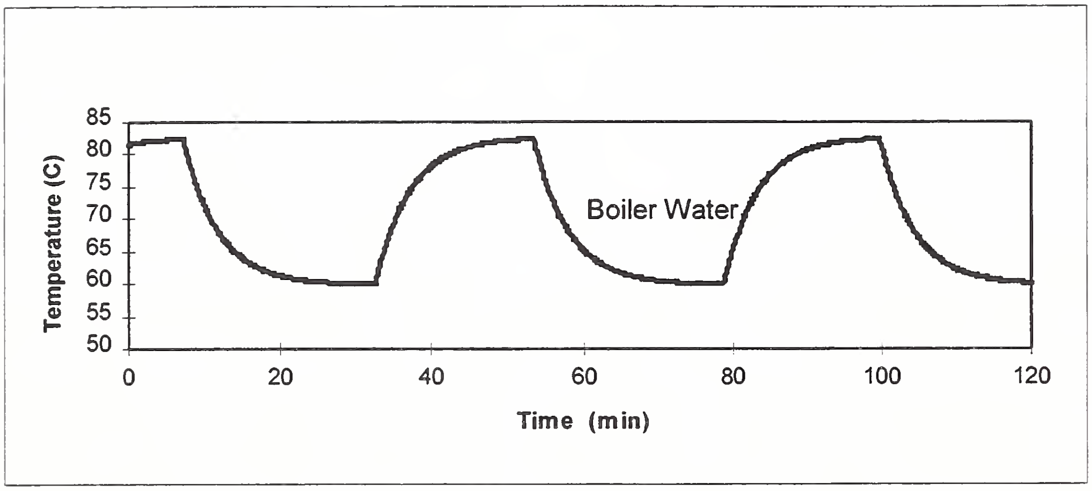

Figure 9. Boiler water temperature.

Table 1 shows the effect of the boiler return water temperatures, T_{rw}, on the cyclic operation. As the return water temperature increases, the period of on/off cycle decreases. The cyclic efficiency, η_{cyc}, is also decreased. The quantities E^*_{inp} and E^*_{out} are the cyclic input energy $E_{input,cyc}$ and output energy $E_{input,cyc}$ normalized by the steady-state input energy per cycle, $E_{inp,ss}$.

Table 1. The effects of return water temperature on boiler cyclic operations.

Trw (C)	Trw(F)	On-time(min)	Off-time(min)	Period(min)	Eff,cyc	Einp*	Eout*
60.00	140.0	21.0	25.4	46.4	0.7955	0.4526	0.3600
65.56	150.0	8.6	23.9	32.5	0.7607	0.2646	0.2013
71.11	160.0	5.2	22.4	27.6	0.7561	0.1884	0.1425
76.67	170.0	3.1	20.2	23.3	0.6808	0.1330	0.0906
79.44	175.0	2.0	18.3	20.3	0.6210	0.0985	0.0612
81.11	178.0	1.2	15.9	17.1	0.5223	0.0702	0.0367
82.11	179.8	0.4	9.8	10.2	0.1577	0.0392	0.0062

Figure 10 gives a good correlation between the cyclic input energy and the cyclic output energy for various circulating water mass flow rates through the boiler. A linear relationship can be observed independent of the water flow rates. The solid linear line is a regression line expressed by

$$E^*_{out} = 0.8218646\,E^*_{inp} - 0.01686\,, \tag{25}$$

with $R^2 = 0.99972$ and the standard error = 0.004152 using the regression equation, the stand-by loss at zero load condition can be estimated to be 2.05 %. When all the data points in Figure 10 are re-plotted for the cyclic efficiency η_{cyc} versus the normalized cyclic input energy, the efficiency curve in Figure 11 is obtained, where the solid line is the regression line of data points.

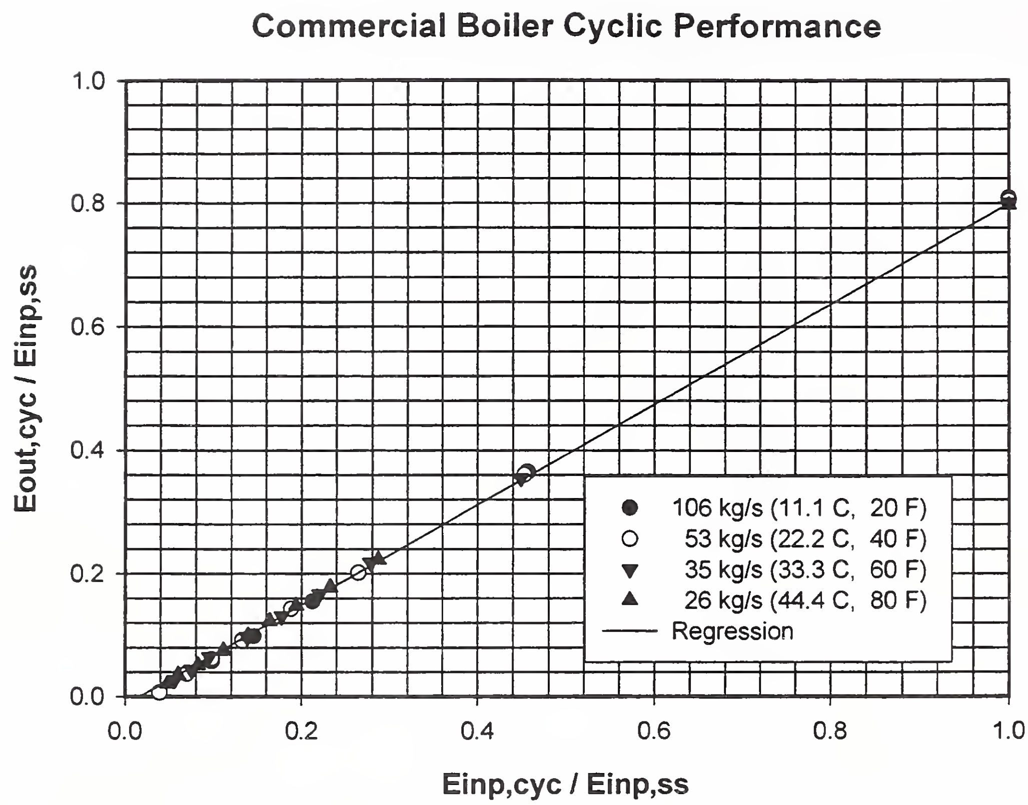

Figure 10. The correlation between cyclic input energy and cyclic output energy for various circulating water flow rates.

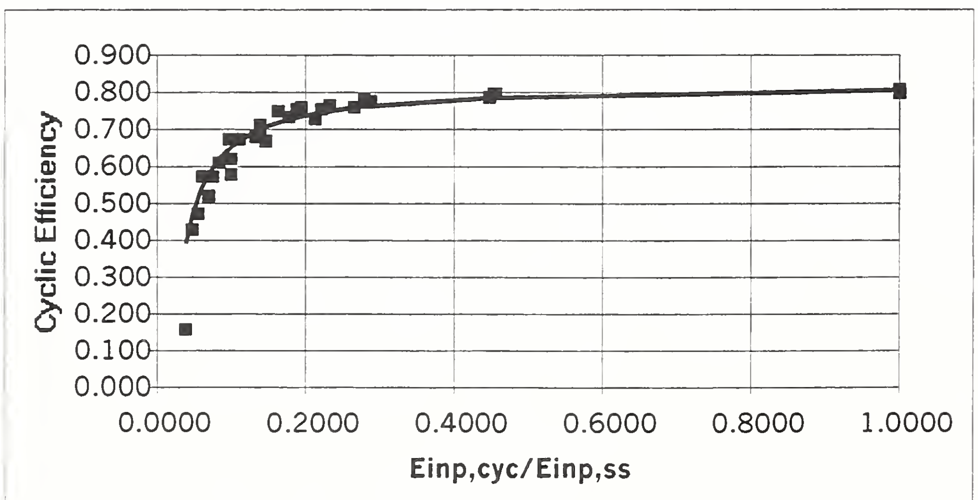

Figure 11. Cyclic fuel utilization efficiency with respect to cyclic input energy normalized by steady-state input energy.

5. Suggested Simplified Part-load Performance Evaluation Method

Based upon the Figure 10, part-load performance of commercial boilers could be evaluated by a simple method. The suggested method is as follows:

(1) Obtain the steady-state output energy experimentally,
(2) Obtain the stand-by loss energy at zero load experimentally,
(3) Draw a straight line connecting the two points, and
(4) Use this line to determine part-load output at different load conditions.

The method is suggested as an approximate method. The errors resulting from use of the method need to be investigated for a wide range of boiler design and sizes.

6. Summary

A commercial steel fire-tube hot water boiler was simulated to obtain steady state and part-load performances using a computer simulation program. A good linear correlation between the cyclic input energy and the cyclic output energy for various circulating water mass flow rates and corresponding temperature rises through the boiler was obtained. A simple method for determining part-load performance of commercial boilers is presented. Experimental validation for the proposed method has not been established yet.

7. References

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